

Wave Loading

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Wave Loading

Introduction

1 The loads caused by waves on surface ships are by far the most important in normal service and will dominate the design of the primary structure. The presence of a hull in the sea will result in a general lack of correspondence between mass distribution ($m(x)$) and buoyancy distribution ($b(x)$) along the length. If a ship was able to maintain an equilibrium position on a fixed wave profile then the total mass and mass moment must be equivalent to the buoyancy and buoyancy moment, that is:

$$(3.1) \quad \int_{-L/2}^{L/2} (m(x) - b(x)) dx = 0$$

$$(3.2) \quad \int_{-L/2}^{L/2} x (m(x) - b(x)) dx = 0$$

where L is the waterline length and x is measured forward from amidships. Then at any given section x_i there will be a bending moment given by:

$$(3.3) \quad \int_{-L/2}^{x_i} (x + L/2) (m(x) - b(x)) dx = M_i$$

The solution of these equations for a given sea surface profile is known as quasi-static balance. Due to the dynamic effects of the motion of the ship this equilibrium is, of course, never achieved except with the hull at rest in still water. Consequently, the effects of translational and rotational accelerations, as well as the instantaneous lack of static balance, need to be added to the quasi-static calculation. This is a complex task and is discussed in detail later in this chapter, although the general problem has been known to naval architects for many years.

2 The bending moment in still water is a special case insofar as there are no dynamic considerations and it can be found by deterministic calculation. Although, therefore, not strictly due to wave action, it will nevertheless be covered under the discussion of wave action to save later repetition of what is a standard procedure.

3 The full dynamic analysis of wave loading has to be dealt with in a linear manner because the complete non-linear equations are intractable with currently available computational tools.

The assumption of linearity means that the hull is taken to be wall-sided to the full amplitude of the wave; this is, at any section the buoyancy per unit length is directly proportional to wave elevation. Additionally, the wave surface is assumed to be the summation of a large number of sine waves, with the calculation being carried out for single waves comprising a wave spectrum and the individual responses being summed over all the sine waves within the spectrum. The effect of these assumptions is that the derived bending moment varies linearly with wave height for a given speed and heading, and there is no difference between sagging and hogging responses. Loads are also overestimated because the hull is assumed to be 'infinitely wall-sided' and buoyancy is added or removed even when the wave profile is above the deck or beneath the keel. Ship motions in relatively benign sea states can be estimated with reasonable accuracy, but structural responses in the high sea states of interest to the structural designer are of doubtful validity. (Note that wave *height* is defined as distance from trough to crest, while *amplitude* is the maximum elevation from the still water line.)

4 From the discussion above and in Chapter 2 it is clear that the question of the true magnitude of wave loading on a hull in a given lifetime is a very complex one. It has been approached with increasing sophistication over the years as greater numerical computing power has become available, but to provide a thorough understanding of the current position it is worthwhile to present it in a historical perspective.

History of Wave Bending Prediction

5 The first classical publication on Naval Architecture in the modern sense is by Rankine (1866) entitled 'Shipbuilding, Theoretical and Practical' and which foreshadows much of the developments of the next century. He is the first to define rigorously such terms as stress and strain, and also includes clear descriptions of the strength and mechanics of beams and trusses. His exposition of bending moment and shear force calculation is surprisingly modern and by dividing the ship into two halves for the estimation of bending moment amidships, he provides the basis for the much later 'Murray's method' (paragraph 15).

6 Rankine also presents the bending moment prediction formula:

$$(3.4) \quad M = \Delta L/C$$

later attributed to John (1874), where Δ is displacement and C is a constant suggested as equal to 20 in hogging as an upper bound on M for rectangular section hulls with an assumed standard mass distribution. Sagging at that time is always dismissed as being less than hogging in the short ships of the day, and deck buckling was apparently not a problem as most decks were still of wood even on iron hulls. Rankine also gives tabular methods for obtaining the position of the neutral axis and the second moment of area, thus providing all that is necessary for calculating stresses in the hull as a simple beam.

7 Reed (1871) goes further and presents a number of examples of graphically derived shear force and bending moment curves for various warships, both in still water and balanced on waves.

In fact Reed appears to be first to use balancing on a wave crest or in a trough as explicit worst cases for wave loading. He also presents an intuitive argument for waves equal to the length of the ship giving the worst load.

8 The question of waves and the effect of dynamic loading was being pursued by a number of researchers in the late 19th Century, notably Smith (1883) and Read (1890) who present methods for allowing for the effects on water pressure (and therefore on bending moment) of particle circulation in waves, and of pitch and heave. These methods were developed with sophisticated graphical solutions by Kriloff (1898) and Alexander (1911). However, Robb (1918) shows that the extreme hogging curve for quasi-static balance calculation covers all other conditions and that the Smith and Read corrections have insignificant effect compared with the effect of changes in the mass distribution. Thereafter the use of corrections, especially that due to Read, becomes progressively rarer, although the Smith correction is once again coming into prominence as it may be significant in the extreme waves now used in design load prediction as discussed by Clarke (1986).

9 Continuing with the development of the design process, John (1874) took the method as described by Rankine (1866) and, by applying it to a number of ships, deduced the formula:

$$(3.5) \quad M = \Delta L/35$$

which subsequently became known as the 'Standard Bending Moment Formula' or more lately 'John's Formula'. The particular aim of this formula was to prevent the overstressing of large ships by relating bending moment to length as well as displacement. The Lloyd's Rules which were current at that time required scantlings to be directly proportional to displacement with the result that the factor of safety against yield fell from about 5 for a 100 ton ship to 2 for a 4000 ton vessel. It should be noted that John intended the formula specifically for merchant ships. There was no agreed constant for warships until the work of Champness (1922) confirmed the value of 35 for hogging loads but gave 29 for sagging.

10 While the standard formula gave an estimate of bending moment, there was still a need for an agreed acceptable stress in the deck and keel, and at later stages of the design where static balance was used to estimate the bending moment, there was also a need for an agreed wave height. On the first point there was no common consensus at the time and there appear to be no recorded values of acceptable stresses in any design documentation since. However, the notes provided for students of Naval Architecture at the Royal Naval College, Greenwich in the early 1960s require that the maximum deck compressive stress in a mild steel frigate (of about 100 m length) should be not more than 5.5 tons/sq.in (85 MPa) and at the sketch design (first iteration) stage 4.5 tons/sq.in (70 MPa) should be the limit. The notes also state that higher stresses may be acceptable in higher strength steels, but that in compression the advantage may be limited by buckling effects. Although the values are quoted for a frigate, clearly material properties are independent of ship size (except insofar as properties vary somewhat with material thickness - see Chapter 5) and these values are a good guide for any designer of a large steel structure at the beginning of the design process.

11 On the question of wave height and shape, the first 'modern' statement is by Biles (1893-4) who says that the $L/20$ trochoidal wave is the usual assumption. White (1894) also discusses wave height and form at some length showing how waves in the sea can be approximated to by a trochoid. Implicit in all the discussions is that the worst wave length (λ) is that equal to the length of the ship, which is a reasonable intuitive conclusion. More recent numerical analysis by Smith and Brown (1976) and Bishop and Price (1979), and model tests reported by Brown (1977(a) and 1978) and by Lloyd et al (1980), suggest that the worst case can, under some circumstances, be on a wave slightly shorter than the ship (about $0.9L$) but, for practical purposes, $L/\lambda=1$ gives a reliable design condition.

12 Returning to wave height, White (1894) gives a table of ship designs showing height used varying from $L/15$ to $L/20$ for warships and $L/20$ to $L/30$ for merchant ships but gives no guidance to the reader on what is best. There is no obvious correlation with ship size or type but it is clear that warships were expected to be significantly more heavily loaded than merchant ships. However, the report of the Committee on Torpedo Boat Destroyers (1901) quotes $L/20$ as a requirement and Biles (1908) makes an unequivocal recommendation of the $L/20$ wave; it subsequently gained worldwide acceptance, being still used extensively today. A variety of other wave heights have been suggested over the years and some of them are illustrated in Figure 3.1; probably the most commonly used is $1.1\sqrt{L}$ (feet) or $0.6\sqrt{L}$ (metres) which allows for the reduced probability of a high longer wave being encountered by balancing longer ships on relatively lower waves.

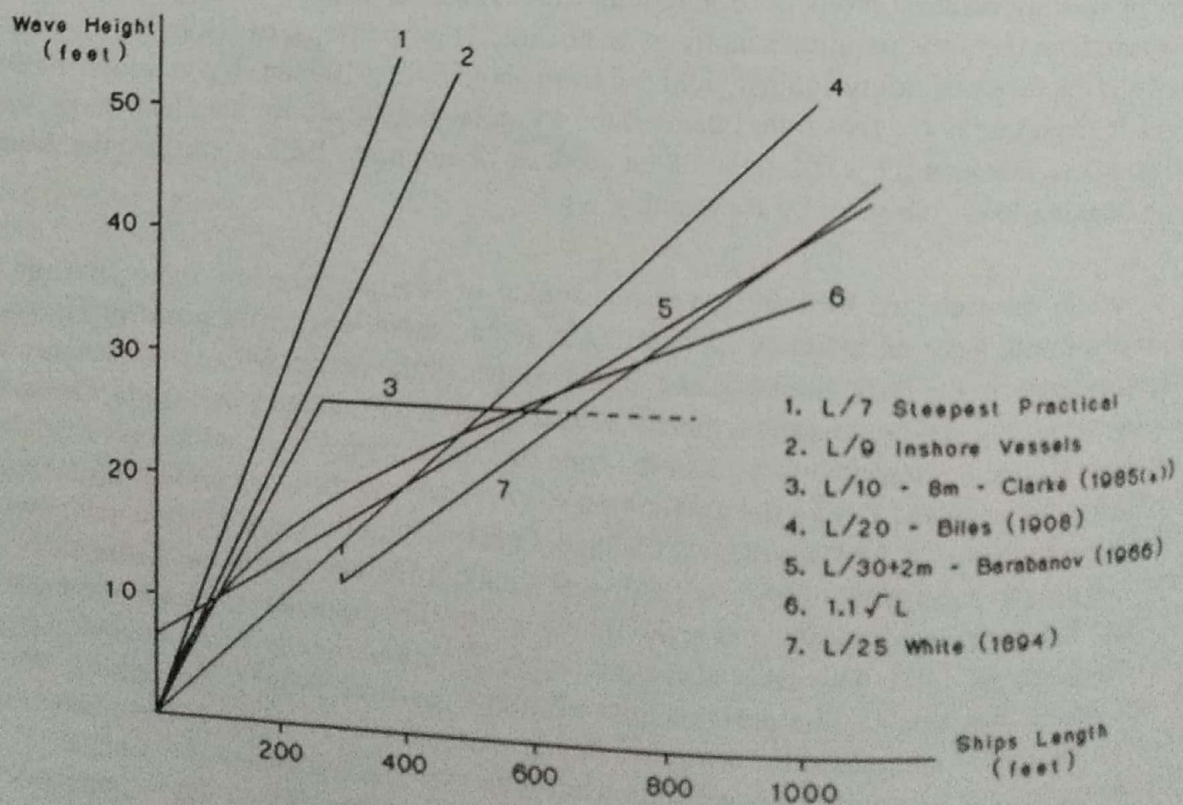


Figure 3.1 Various Proposed Design Wave Heights

13 The initial design process was then to estimate the bending moment from the standard formula, equation 3.5, and using an allowable stress (σ) to derive the necessary section modulus (Z) at amidships from the beam formula:

$$(3.6) \quad Z = M/\sigma$$

A section could then be synthesised by trial and error to match the required modulus, and then scaled lengthwise using experience to give the scantlings for the rest of the hull. The structural weight could then be deduced and added to all the other ship weights which had been estimated by comparable means, giving a complete ship mass distribution. A static balance was then carried out to provide a more accurate estimate of bending moment and the process repeated progressively to refine the scantlings.

14 A variation on this method was proposed by Biles (1908) in which he suggested that for merchant ships a standard mass distribution could be used. That was a trapezium with the same mass and centre of gravity as that required for the ship and which became known as 'Biles' Coffin' (Figure 3.2). It was used extensively to avoid use of the standard formula and to enable entry to the static balance calculation at the start. For warships with a much more uneven mass distribution Biles' method was not appropriate and standard formulae continued to be used. A method for estimating mass distribution for warships based on a constant volumetric density has been proposed recently by Chalmers (1982 and 1988(b)) and will be presented in more detail in Chapter 6.

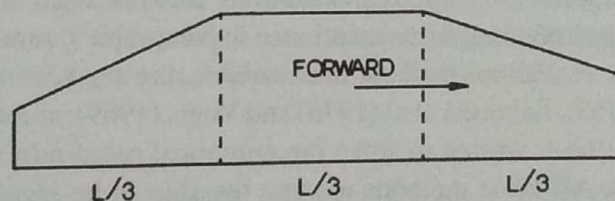


Figure 3.2 Biles' Coffin Mass Distribution

15 Historically, perhaps one of the most important milestones came with Murray (1946-7) who collected a large volume of data and suggested a much more refined method for estimating bending moment in which moments due to mass, still water buoyancy and wave buoyancy are each calculated or estimated independently. The wave bending moment, defined as the difference between the still water condition and that with the ship on a wave, is proportional to L^3B (B is the beam amidships), the still water bending moment is proportional to L (as John's formula) but is also linearly related to block coefficient (C_b), and the moment due to mass is calculated from an assumed known mass distribution. An estimated distribution such as Biles' Coffin may be used if the real distribution is not known, but Murray does not recommend it.

16 Murray's method is still one of the most useful for merchant ship design but unfortunately, presumably due to the much more complex hull form and mass distribution, the method has never been successfully developed for warships. Baker in his discussion to Murray's paper (1946-7)

tries to fit a number of warship designs in Murray's theory but fails to get any reasonable correlation. He suggests that relating the still water bending moment to prismatic coefficient (C_p) rather than block coefficient might be more reliable for warships, but the idea has never been followed up. Murray's curves for still water bending moment do not extend to a low enough value of C_b for warships and simple extrapolation does not seem to give sensible answers: Lloyd's rules, where they quote the method, say that it should not be used for values of C_b less than 0.6.

17 At about that time the use of statistics began to be proposed for the prediction of design bending moments. A number of publications appeared through the 1950s and 1960s of which Turnbull (1953), Lewis (1957), Telfer (1960), Yuille (1963), Ward (1963), Murray (1963), Moor (1967) and Murdey (1972) are all worthy of study as they increase understanding of the subject, although none have had any lasting effect on commonly used design processes. The text books written in the same period, in particular Barabanov (1967), Muckle (1967), Harvey Evans (1975) and Comstock (1977) all still advocate the use of John's (1874) method or Murray's (1946-7) method for initial estimation of bending moment; the use of the simple beam formula and an allowable stress to give an acceptable section modulus and static balance on, usually, a $1.1\sqrt{L}$ ($0.6\sqrt{L}$ metric) wave for subsequent iterations.

18 In recent years a considerable amount of work has been put into the development of numerical methods for derivation of design bending moments so as to be able to allow for actual wave form, relative ship motions in waves and the dynamic response of the ship, rather than a notional quasi-static condition. These methods are known generically as **strip theory**; the ship is divided into vertical strips and the linear response of each strip to a sinusoidal wave excitation is calculated while satisfying the compatibility conditions between each strip. For a complex or irregular sea linear superposition is used, as mentioned in paragraph 3, summing the sine waves to achieve the desired sea state. Various methods are available, the most commonly used ones being due to Gerritsma et al (1967), Salvesen et al (1970) and Vugts (1969), and a number of equivalent computer programs have been written to solve the numerical equations, for example SCORES reported by Raff (1972). All these methods assume the ship to be *rigid* beam; more recently Bishop and Price (1977) have developed a flexible beam strip theory which takes account of bending and shear stiffness of the hull when solving for compatibility between strips. This type of theory is the only one currently openly available which can estimate the distortional higher frequency responses of a hull to slamming and whipping excitation, but it is still essentially a linear analysis and extreme effects are not well modelled. The detail of rigid and flexible beam strip theory is discussed later (paragraphs 31-49). Efforts have been made to extend strip theory into the non-linear regime, for example Jensen and Pedersen (1979), but they remain mainly of academic interest.

Current Available Prediction Methods

19 From the foregoing historical survey it is clear that the modern designer has to choose between traditional methods of bending moment estimation going back 100 years and more and not allowing explicitly for the dynamic effects of ship motion in waves, or embracing more modern computer based analyses which allow dynamic response but lose the effects of non-

linearities in the hull shape and the wave excitation. In considering the choice it is necessary to look at the uncertainties in the methods more closely and to compare the results of analyses with measurements taken from ships in service. These latter data are, after all, the only hard information available to the designer about what it actually happening at sea.

20 The accuracy of static balance calculations in relation to the need or otherwise to allow for dynamic effects was investigated by Robb (1918) who showed that the standard calculation without any allowance for heave and pitch motions was conservative, and that dynamic effects reduced the estimated bending moments. However, his work related to the relatively low, though severe, waves around the $L/20$ height at which hogging bending moments are very similar to sagging ones; indeed, for the ships of that time hogging was seen as the worst case. In the extreme conditions which relate to the **Ultimate Limit State**, that is the point at which the hull will fail irreversibly (contrasting with the **Serviceability Limit State** when the capability of the hull starts to be degraded, but may still be safe), the worst condition is sagging as illustrated in *Figure 3.3*. This shows, for static balance on increasing wave heights, that the hogging bending moment falls below the linear relationship which would be expected for a wall-sided hull, at a much lower wave height than the sagging response. Measurements from sea presented by Brown (1977(b) and 1981) and Clarke (1985) and model tests presented by Brown (1978) and Lloyd et al (1980) confirm this conclusion for real conditions. It is clearly the extreme conditions which will damage or sink ships, and for the designer to have an understanding of the margins he has between design loads and failure so that he can obtain the optimum performance for his structure, it is necessary to investigate whether Robb's conclusions are equally valid in the extreme sagging state.

21 A study of Robb's (1918) paper shows that the sagging case, although of lesser magnitude than hogging, is significantly affected by heave and pitch. Indeed, Robb himself notes the fact and remarks that for ships of high speed form such as liners or ferries (and therefore including

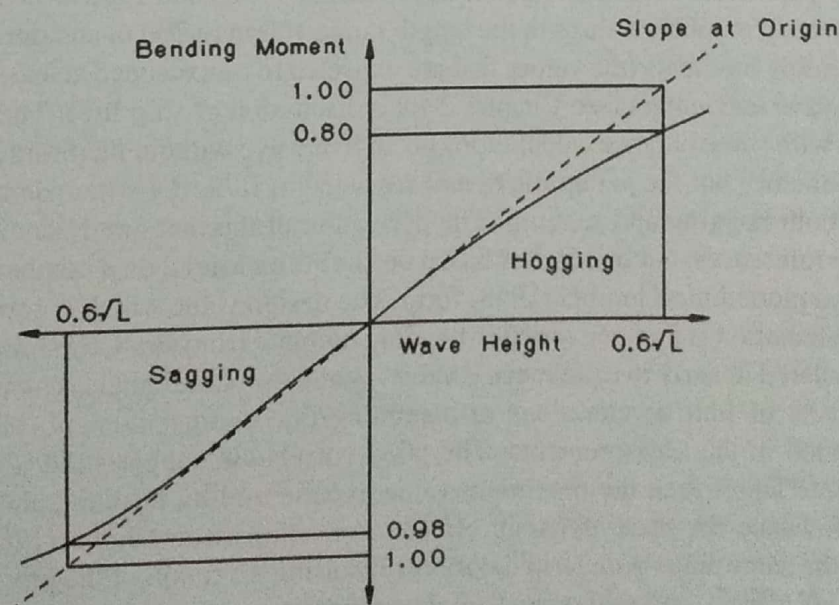


Figure 3.3 Comparison Between Extreme Hogging and Sagging Bending Moments

warships) his conclusions may not be valid, especially if the vessel has heavy machinery amidships increasing the still water load. Unfortunately, as noted earlier (paragraphs 3 and 18), strip theory gives no help in such comparisons as the hull form is assumed wall-sided and hogging and sagging responses are identical. In the absence of any non-linear theory, which would in any case be numerically extremely complex and need to operate in the time domain (as opposed to the frequency domain for linear strip theory) the best solution to the problem is to compare static balance results with measurements taken from ships at sea, and to study the correlation.

TABLE 3.1

Comparison Between Expected Maximum Bending Moment Based on Measured Strains and Static Balance on a Wave of Height $H = 8$ m

Ship Class	H/L	BM from Static Balance (MNm)		Ratio Measured/Calc	
		Hog	Sag	Hog	Sag
Type 12 (Rothesay)	0.073	122	180	1.02	0.97
Type 81 (Tribal)	0.075	111	160	0.98	1.14
Leander (N Beam)	0.073	123	177	1.12	1.17
Leander (B Beam)	0.073	127	194	0.97	1.11
County	0.052	325	425	1.00	1.05
Type 21	0.073	139	221	1.08	1.11
Type 42 (Batch 1)	0.067	165	264	0.81	0.90
Type 22 (Batch 1)	0.064	188	282	0.90	0.97
CVSA	0.042	2200*		0.96*	
Average (not including CVSA)				0.98	1.05
* Total range				±0.10	±0.10

22 The only published work in this area is due to Clarke (1985 and 1986) who has taken strain measurements from a number of ships in the length range 100 m to 200 m and derived from them the expected bending moments (the values that are expected to be exceeded at least once) in a ship life of 3×10^7 wave encounters (see Chapter 2 for a discussion of ship life). These results have been compared with static balance calculations on an 8 m wave without the Smith correction (for numerical convenience but see paragraph 8) and are listed in Table 3.1 which demonstrates good correlation for both hogging and sagging. The derivation of the measured bending moments is explained in the references, but briefly it is based on strains measured on a number of ships of the classes listed and plotted in a Gumbel (1958) form. The design value which is compared with the static balance calculation is first converted to bending moment from strain by simple beam theory and then extrapolated linearly to the expected value. While the static balance calculation does not include the effects of hull accelerations or slamming on bending moment, these effects are implicitly included in the measurements. The good correlation suggests that the design wave height is therefore larger than the maximum value experienced by the ship, and the difference provides an allowance for these dynamic effects. For comparison Clarke (1985) reports that correlation for the same ships with strip theory results using a version of the program SCORES, described by Raff (1972), and with statistical data taken from ships' logs for the probability of different speeds and headings in different sea states yields a measured/calculated bias of $1.03 (\pm 0.10)$ for hogging but $1.22 (\pm 0.13)$ for sagging. The difference between hogging and sagging in

this strip theory calculation is achieved by scaling from static balance results but even so the sagging prediction does not correlate as well as the results presented in *Table 3.1*.

23 However, strip theory does take account of phenomena which cannot be explicitly included in static balance bending moment predictions and it may therefore be useful for providing an insight into unusual happenings resulting from dynamic responses which are related to non-extreme and so relatively linear motions, for example the effects of different modes of operation on fatigue loading, while it may appear therefore that a version of static balance is currently the most accurate prediction method, neither method is particularly good at predicting extreme load values, and strip theory undoubtedly has more scope for development. Both methods are consequently presented in the following paragraphs, in each case with a procedure for extrapolating the result to an extreme condition appropriate to the design value, that is the ultimate limit state.

Design Bending Moment by Static Balance

24 Static balance theory is straightforward and may be found in many Naval Architecture text books, for example Rawson and Tupper (1976) or Comstock (1967), and in any case the method is now embodied in well established and validated computer programs. It will be assumed, therefore, that the designer is able to carry out a static balance calculation on any height of trochoidal wave for a conventional monohull structure. The aim then is to achieve a resulting bending moment which reflects an expected probability of exceedance in a given number of wave encounters. As discussed above (paragraph 22) for warships in the length range 100 to 200 metres, the expected maximum bending moment in 3×10^7 wave encounters for British naval areas of operations can be approximated to by balancing the ship on an 8 metre wave of the same length as the ship. However, this is an expected value, not a design value, and there are other uncertainties implied by the standard deviations in *Table 3.1*. Furthermore, the strain measurements from which the 8 m wave height was deduced were taken from ships with a worldwide operating history. The conditions in the North Atlantic are such that ships of around frigate and destroyer size meet waves of their own length with a greater frequency than in most other sea areas. There is evidence that ships in this length range (100 to 150 metres) could, for the same probability of exceedance, be subjected to lifetime design bending moments up to 15% higher than the worldwide average if operating *solely* in the North Atlantic.

25 A suggested procedure for estimating design bending moments is probably best illustrated by an example and the designer can insert his own parameter values where those differ from the following. The only critical parameter which needs to be decided and on which most of the rest depends is the required probability of exceedance of the design bending moment in a ship life. A study of failure rates in other engineering disciplines, see for example Brown and Chalmers (1990), particularly those associated with large steel structures, suggests that a figure of the order of 1% is about average, and that is what will be taken here as an example. It is therefore a question of investigating the uncertainties associated with the 8 m static wave balance, and then extrapolating the result from the expected value to that which has a 1% probability of exceedance in 3×10^7 wave encounters. For other numbers of wave encounters see paragraph 30a.

26 The systematic errors associated with the measurements against which the calculation is calibrated are:

- Mean bias against the 8 m wave balance of 1.02 with standard deviation of 0.10 (Table 3.1).
- A consistent underestimate of strain from the mechanical strain gauges of about 5% due to friction, discussed by Clarke (1986).
- Inaccuracies in the calculation of the hull section modulus for converting strain to bending moment with a mean of about 5%, discussed by Chalmers (1988(b)), and a scatter which will be included in the standard deviation of (a). (There is, however, some evidence that this value of 5% may be a little optimistic as discussed by Brown (1977(b) and 1981).

These errors may be assumed to be independent and so combining them gives a bias of 1.12 with a standard deviation of 0.10. Applying these factors to the static balance result would then give the expected bending moment in a ship's life which, assuming a normal distribution, has a 63% probability of exceedance.

27 The requirement is for a 1% probability of exceedance. A typical Gumbel plot of measured values is shown in Figure 3.4 which is extrapolated linearly from the expected point to

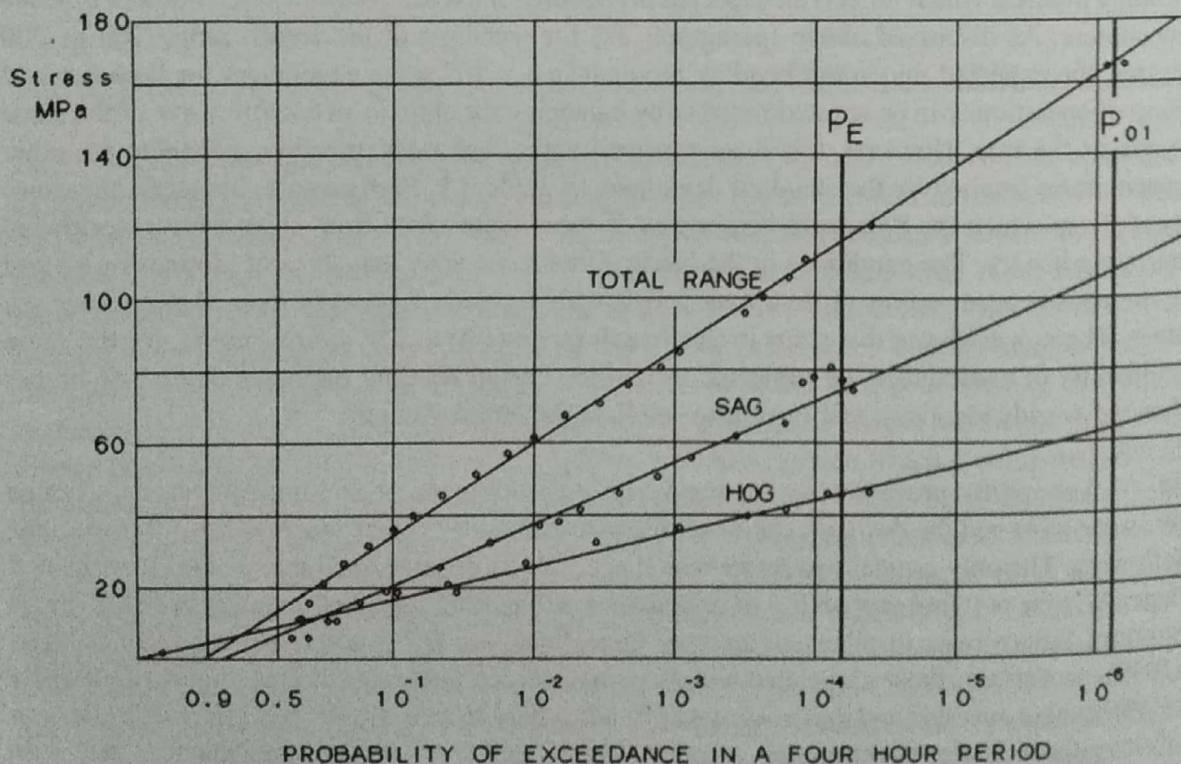


Figure 3.4 Gumbel Plot of Typical Deck Stress Measurements

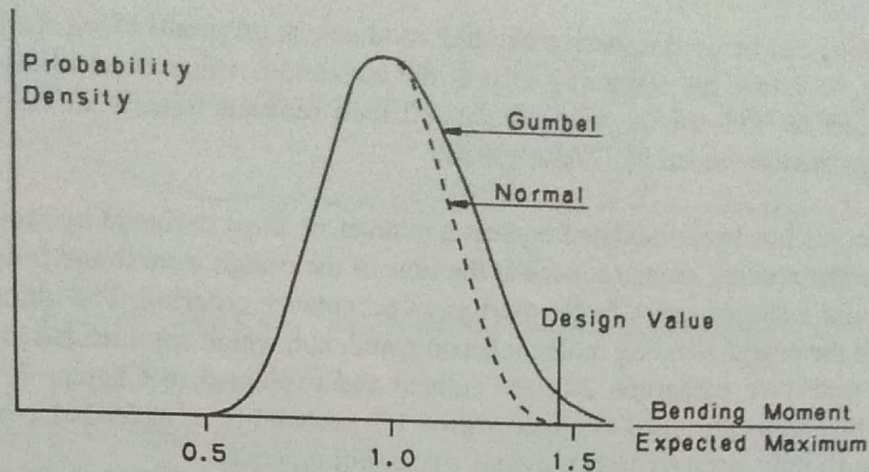


Figure 3.5 Probability Density Distribution of Maximum Hull Girder Bending Moment

the 1% point. As is described by Clarke (1986), the actual measurements above the expected point, insofar as they exist, tend to drop below the straight line due to non-linearities in the extreme response. The effect is to change the distribution about the expected value from a skewed one to an approximation to a normal one as shown in *Figure 3.5*. The standard deviation of this 'quasi-normal' distribution is 0.15. Combining this deviation with that above, assuming they are independent, gives a value of 0.18. Note that, as explained in paragraph 24, a worldwide average design bending moment is still being estimated. A further partial factor will be needed if a significant proportion of the ship's life is to be spent in the North Atlantic.

28 To achieve a 1% probability of exceedance 2.33 standard deviations are needed and so combining this with the bias gives a factor $1.12 + (2.33 \times 0.18) = 1.54$. This factor only applies to the wave bending component of the static balance results because the measurements on which the correlation is based are zeroed at the still water condition. It may be assumed that the still water bending moment can be calculated directly with no significant error, although there will clearly be some scatter due to uncertainties in the displacement and mass distribution and the fact that there will be some variation with ship speed, as shown by Brown (1978) and Lloyd et al (1984). However, still water bending moment is only about 10% of the wave bending moment and so any inaccuracies will have relatively little effect. Nevertheless, the designer should try to estimate the worst still water bending moment to which the ship is likely to be subjected over its life.

29 **Design bending moments** may then be deduced from the following formulae:

$$(3.7) \quad M_{ds} = M_{sw} + 1.54 (M_s - M_{sw})$$

$$(3.8) \quad M_{dh} = M_{sw} + 1.54 (M_h - M_{sw})$$

where M_{ds} , M_{dh} are the design sagging and hogging bending moments;
 M_s , M_h are the sagging and hogging bending moments from static balance on an 8 m wave;
 M_{sw} is the still water bending moment making proper allowance for its sign (usually +ve for hogging and -ve for sagging).

These expressions can be used to derive bending moments at all points along the length of the ship. However, to allow for slamming effects the maximum value at amidships should be extended forward by 15% of the ship length and then reduced linearly to zero at the fore perpendicular as recommended by Clarke (1986).

30 This process has been validated against a number of ships designed by more traditional methods where the bending moments used at the time of the design were found from balance on an $L/20$ wave and a simple stress factor used as an acceptance criterion. The ultimate strength values to which the design bending moment is compared, and which are intended to relate to the ultimate limit state (see paragraph 20), are defined and explained in Chapter 7. The overall implication is that ships designed to a load with a 1% probability of exceedance by the above method will be slightly stronger than previous successful practice.

31 The number of 3×10^7 wave encounters was chosen to reflect a typical RN warship life as indicated in Chapter 2 (paragraph 10). If however the ship is to be designed for a longer or more intensive life then 3×10^7 wave encounters will be insufficient. The logically correct way to allow for more wave encounters would be to increase the static balance wave height, but to achieve this would mean repeating the previous work with different parameters. A simpler and sufficiently accurate method is to increase the 1.54 factor in equations 3.7 and 3.8. This can be done by using the process of Clarke (1985) to derive the 1% exceedance probability for longer lives and then using the slopes of the lines in *Fig. 3.4* to provide a ratio for longer lives. Using this derived ratio the factor to be used in equations 3.7 and 3.8 is then as shown in *Table 3.2*.

TABLE 3.2
Bending Moment Factor for Higher Numbers of Wave Encounters

No. of wave encounters	Factor in equations 3.7 & 3.8
3×10^7	1.54
4×10^7	1.57
5×10^7	1.59
7×10^7	1.63
10×10^7	1.67

It is possible that these factors are a little optimistic due to the greater uncertainty implicit in the further extrapolation, but this should be offset by the known trend for stresses to fall below the linear extrapolation at lower probabilities.

Strip Theory

32 There is no particularly clear explanation of strip theory available and probably only that due to Bishop and Price (1979) is readily accessible. Consequently, a brief summary of the method will be given here. As the loading response of the hull is a function of both the inertial and fluid forces, and these are functions of the ship motion, it is first necessary to solve the rigid body equations of motion.

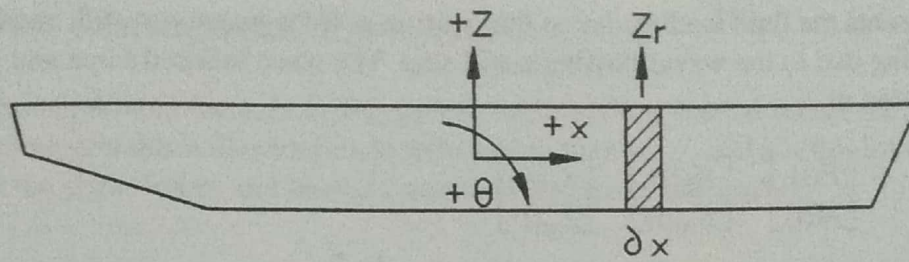


Figure 3.6 Strip Theory - Axis System and Nomenclature and Sign Convention

33 For the axis system shown in Figure 3.6 the following are the equations of motion for heave and pitch, neglecting, for simplicity of presentation, the damping terms $\dot{z}$ and $\dot{\theta}$:

$$(3.9) \quad \begin{bmatrix} \Delta \ddot{z} \\ I_L \ddot{\theta} \end{bmatrix} = \begin{bmatrix} F(t) \\ M(t) \end{bmatrix}$$

where $\Delta =$ mass displacement

$I_L =$ pitch inertia

$F(t)$, $M(t)$ represent the wave induced force and moment acting on the ship and are determined by integration of the wave induced fluid loading $\frac{\partial F}{\partial x}(x, t)$ as indicated in Figure 3.6. Note that fluid damping effects are included on the *load* side of the equation, as stated in paragraph 35, to simplify the solution so that

$$(3.10) \quad F(t) = \int_L \frac{\partial F(x, t)}{\partial x} dx \text{ and } M(t) = \int_L \frac{\partial F(x, t)}{\partial x} x dx$$

34 In elementary strip theory it is assumed that the wave induced fluid loading for a strip at x is a function of the relative vertical displacement $z_r(x, t)$ such that

$$(3.11) \quad z_r(x, t) = z(x, t) - \zeta(x, t)$$

where z is the absolute displacement and ζ is the wave elevation at x (+ve up).

35 It is further assumed that the fluid loading is made up of three terms, the fluid added mass, fluid damping and the effect of buoyancy. The first two terms are functions of strip shape, encounter frequency and relative vertical displacement and details of their computation can be found in Bishop and Price (1979). The third term is simply $\rho g B(x) z_r(x, t)$ where $B(x)$ is the (constant) beam at x .

36 The fluid force may then be re-expressed as functions of z and ζ so that

$$(3.12) \quad \frac{\partial F}{\partial x} = \frac{\partial J}{\partial x} + \frac{\partial K}{\partial x}$$

where J represents the fluid loading due to ship motion as if the water was still, and K represents the fluid loading due to the waves passing a still ship. The wave induced force and moment can then be rewritten as

$$(3.12) \quad \begin{bmatrix} F(t) \\ M(t) \end{bmatrix} = \begin{bmatrix} K_F(t) \\ K_M(t) \end{bmatrix} - \begin{bmatrix} J_F(t) \\ J_M(t) \end{bmatrix}$$

where $K_F(t) = \int_L \frac{\partial K(x,t)}{\partial x} dx$ etc, similar to equation 3.10.

37 As

$$(3.13) \quad z(x,t) = z(t) - x\vartheta(t)$$

the terms J_F and J_M can be expressed in terms of z , dz/dt , d^2z/dt^2 and ϑ , $d\vartheta/dt$, $d^2\vartheta/dt^2$ with coefficients depending only on the geometry of the strip and wave encounter frequency ω_e .

38 From equations 3.9 and 3.12 the equations of motion can be written as

$$(3.14) \quad \begin{bmatrix} \Delta \ddot{z} \\ I_L \ddot{\vartheta} \end{bmatrix} + \begin{bmatrix} J_F(t) \\ J_M(t) \end{bmatrix} = \begin{bmatrix} K_F(t) \\ K_M(t) \end{bmatrix} = \mathbf{K}(t)$$

For regular waves the forcing function $\mathbf{K}(t)$ will be of the form

$$(3.15) \quad \mathbf{K}(t) = \mathbb{R} [K \exp(i\omega_e t)]$$

and, as it may be assumed that the displacements are harmonic at the same frequency, then

$$(3.16) \quad \mathbf{D} \begin{bmatrix} z(t) \\ \vartheta(t) \end{bmatrix} = \begin{bmatrix} K_F \\ K_M \end{bmatrix} \exp(i\omega_e t)$$

where $\mathbf{D}$ is a 2×2 matrix of coefficients again depending on strip geometry and ω_e is taken from the left hand side of equation 3.14, hence

$$(3.17) \quad \begin{bmatrix} z(t) \\ \vartheta(t) \end{bmatrix} = \mathbf{D}^{-1} \begin{bmatrix} K_F \\ K_M \end{bmatrix} \exp(i\omega_e t)$$

so that both heave and pitch are harmonic functions with a phase difference relative to the forcing function and so, in regular seas, the heave and pitch response can be estimated for a given value of ω_e from equation 3.17. For irregular seas, if the elevation of the surface is described as a summation of a number of sine waves, the motion responses can be calculated for each sine wave and summed to give the total response. The same process may also be used for the other degrees of freedom, roll, yaw, surge and sway, but for each of these freedoms an additional equation of motion has to be formulated.

Regular Wave Loading from Strip Theory

39 For the whole ship the sum of the applied forces and the inertial forces must be zero, but for an individual strip this will not be so, as indicated in *Figure 3.7*, and it is this imbalance which gives rise to the shear forces and bending moments along the hull girder.

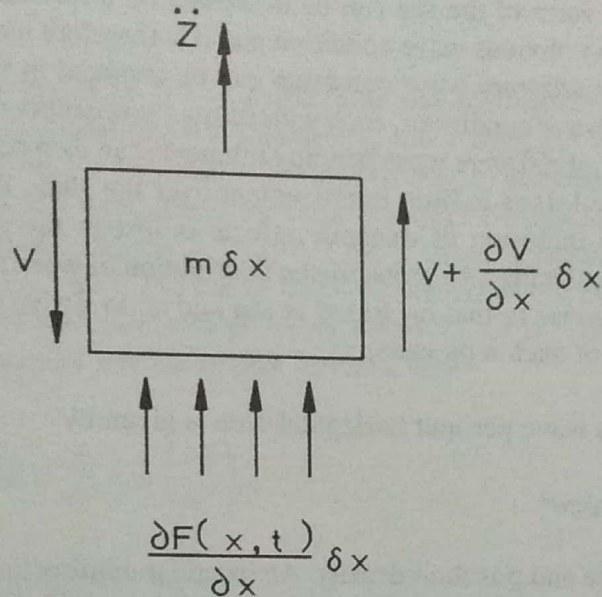


Figure 3.7 Forces on an Individual Strip

40 The equation of motion of the strip is

$$(3.18) \quad (m \delta x) \ddot{z}(x, t) = \frac{\partial V}{\partial x} \delta x + \frac{\partial F}{\partial x} \delta x$$

and hence

$$(3.19) \quad \frac{\partial V}{\partial x}(x, t) = m \ddot{z}(x, t) - \frac{\partial F}{\partial x}(x, t)$$

As z and F are both known functions of z , ϑ , ζ and ω_e which are all harmonic, $\frac{\partial V}{\partial x}(x, t)$ is also harmonic and the shear force (V_j) and bending moment (M_j) at any section x_j can be found by integration as follows, where ϕ_v and ϕ_M are the phase shifts for shear force and bending moment relative to the forcing function.

$$(3.20) \quad V_j = \int_{x_j}^{L/2} \frac{\partial V}{\partial x}(x, t) dx, \quad M_j = \int_{x_j}^{L/2} (x - x_j) \frac{\partial V}{\partial x}(x, t) dx$$

so

$$(3.21) \quad V_j = V(x_j) \exp \{i(\omega_e t + \phi_v)\}, \quad M_j = M(x_j) \exp \{i(\omega_e t + \phi_M)\}$$

Design Bending Moment by Strip Theory

41 It has been shown in the above paragraphs how a value of bending moment response can be deduced for regular sinusoidal wave excitation in head seas, and it has also been suggested how this theory can be extended to allow for irregular seas and different headings. However, all this presupposes that the form of the sea can be described by a summation of sine waves. For design purposes there is no obvious wave condition and it is therefore necessary to do one of two things. Either a particular arbitrary wave condition can be assumed in the same way as it is for static balance, or a set of wave conditions, each with its own probability of being met, can be laid down, and the probability of different wave bending moments can then be deduced. If an arbitrary single sea state is assumed there is little improvement over the static balance method because what is gained from the inclusion of dynamic effects is lost in the assumption of linearity. Consequently it is usual to decide on a probabilistic distribution of wave spectra and to attempt to match the calculated response to that measured at sea and so to derive correlation factors. The following is an example of such a procedure.

42 The energy E in a wave per unit horizontal area is given by

$$(3.22) \quad E = \frac{1}{2} \rho g a^2$$

where a is wave amplitude and ρ is fluid density. An irregular unidirectional wave profile can be considered as being made up of a number of sinusoidal waves of different frequencies and amplitudes each with its own contribution to the total energy, so that an energy distribution

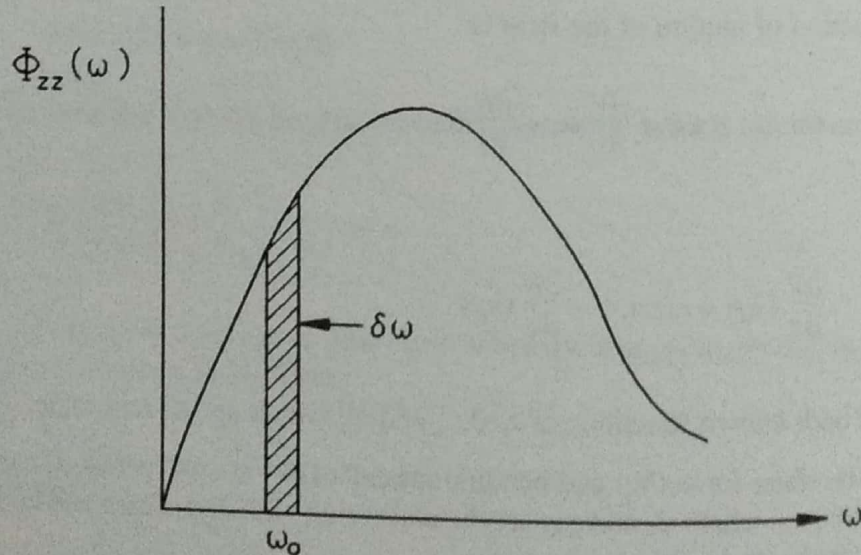


Figure 3.8 Energy Spectrum for Unidirectional Waves

spectrum can be postulated as in *Figure 3.8*. An elemental strip of width $\delta\omega$ centred on frequency ω_0 can be considered to have amplitude a_0 given by

$$(3.23) \quad \Phi_{zz}(\omega_0) d\omega = \frac{1}{2} a_0^2$$

It also follows that the area under the distribution curve is equal to the mean square value m_o such that

$$(3.24) \quad m_o = \int_0^{\infty} \Phi_{zz}(\omega) d\omega = \sum_{r=0}^{\infty} \frac{1}{2} a_r^2$$

43 From lengthy observations, for example those due to Hogben et al (1986), the shape of wave spectra can be correlated to two parameters, the significant wave height ($H_{1/3}$) and a characteristic period (T). $H_{1/3}$ is the mean of the highest third of all observed waves and is equal to $3.8\sqrt{m_o}$ (a little lower than $4.0\sqrt{m_o}$ given by a true Rayleigh distribution; see International Ship Structures Congress (ISSC) Report 1.1 (1982) for a more detailed explanation). $\Phi_{zz}(\omega)$ is therefore a function of $H_{1/3}$ and T , and can be found from a number of empirical formulae, the one most commonly used in structural design being the ISSC, ITTC (International Towing Tank Conference) or Bretschneider two parameter formula, thus

$$(3.25) \quad \Phi_{zz}(\omega) = \frac{A}{\omega^5} \exp\left(\frac{-B}{\omega^4}\right)$$

where $A = (173 H_{1/3}^2)/T_1^4$ and $B = 691/T_1^4$

There is also available the JONSWAP (Joint North Sea Wave Project) spectrum for limited fetch conditions, most appropriate for ships designed exclusively for restricted waters. This spectrum is defined as

$$(3.26) \quad \Phi_J(\omega) = 0.658 C \Phi_{zz}(\omega)$$

where $C = 3.3^q$, $q = \exp [(-1/\gamma^2) \{(\omega T_o/2\pi) - 1\}^2]$ and $\gamma = 0.07$ if $\omega < 2\pi/T_o$ or 0.09 if $\omega > 2\pi/T_o$. The characteristic period T can be described in a variety of ways and for the above constants in the formulae for A and B the characteristic mean period T_1 is assumed but in the JONSWAP formula the modal period T_o is used. Wave statistics may be quoted against a variety of different periods which may be converted to T_1 or T_o as follows taken from ISSC Report I.1 (1979) and Brown (1981):

Modal period T_o (or T_p)	$T_1 = 0.77 T_o$	or $T_o = 1.29 T_1$
Zero upcrossing period T_z	$T_1 = 1.098 T_z$	or $T_o = 1.41 T_z$
Significant period $T_{1/3}$	$T_1 = 0.86 T_{1/3}$	or $T_o = 1.11 T_{1/3}$

These relationships are averaged over a wide scatter of measurements taken from seas in many parts of the world so that, if a spectrum is to be created for a known sea area, reference should be made to the source documents. However, of all the different periods quoted, that which is nearest to the visual period (sometimes called T_v) and therefore that which will be most reliable where the data has been collected visually, is the zero upcrossing period.

44 The spectrum described above, although irregular, is unidirectional and it is necessary to take account of the directional nature of a confused sea. The simplest, and most usual, representation of a direction wave spectrum is

$$(3.27) \quad \Phi(\omega, \chi) = \Phi(\omega) \cdot \Xi(\chi)$$

where $\Xi(\chi)$ is a function describing the spread of the waves about the dominant direction. For the two parameter spectrum quoted in paragraph 43

$$(3.28) \quad \Xi(\chi) = \frac{2}{\pi} \cos^2 \chi$$

is recommended. Further discussion on wave spreading can be found in Bishop and Price (1979).

45 For any particular combination of speed, heading and encounter frequency the response of the ship can be derived from the wave input through the **Response Amplitude Operator**, which is the square of the **Transfer Function** ($H_{rz}(V, \chi, \omega_e)$). These response amplitude operators may be derived theoretically as described above for regular seas and for a linearised hull form, or may be found from model tests. A good description of the process is given by Comstock (1967). Thus, for a given regular wave, the response

$$(3.29) \quad R_i(V, \chi, \omega_e)^2 = |H_{rz}(V, \chi, \omega_e)|^2 a_i^2$$

and it follows that the response spectrum is given by

$$(3.30) \quad \Phi_{rr}(V, \chi, \omega_e) = |H_{rz}(V, \chi, \omega_e)|^2 \Phi_{zz}(\omega_e) \Xi(\chi)$$

The response spectrum may be in terms of motions (heave, pitch) or, for structural purposes in terms of shear force and bending moment, depending on which response amplitude operators are used. The total energy in the response spectrum, m_{or} , can be found by integration as follows and is then used in the statistics of the response.

$$(3.31) \quad m_{or}(V) = \int_0^\infty \int_{-\pi}^\pi \Phi_{rr}(V, \chi, \omega_e) d\chi d\omega_e$$

46 If the bending moments experienced by a ship over a short period of time during which conditions remain sensibly constant are measured at random or at equal time intervals, the statistical distribution of the measurements will approximate to a Gaussian (normal) distribution. However, for most structural purposes the statistical distribution of the maxima and minima is of more interest and for a narrow banded spectrum (which is usual for short term responses) the amplitudes of the turning values follow a Rayleigh distribution (see Price and Bishop (1974)); this form of distribution is confirmed by measurements from sea as discussed by Clarke (1986). This conclusion is convenient because the distribution is a function of a single parameter which can be shown to be equal to the integral of the response energy spectrum, the mean square

response m_{or} . Thus the probability density function for the maximum bending moment B , measured over a short period of time in which conditions remain constant, is given by

$$(3.32) \quad P(B) = \frac{B}{m_{or}} \exp \left[\frac{-B^2}{2m_{or}} \right]$$

The probability distribution function, or cumulative probability, of the maximum bending moment equalling or exceeding a given value of B in any one cycle under the specified conditions is given by

$$(3.33) \quad P(B \geq B) = \exp(-B^2/2m_{or})$$

It can also be shown that the probability of exceeding the response B at least once in N cycles is given by

$$(3.34) \quad P_N(B \geq B) = 1 - \exp \{-N \exp(-B^2/2m_{or})\}$$

47 The short term statistics described above only remain valid as long as conditions remain steady. At sea this is unlikely to be for more than about an hour and so it is necessary to take account of the probability of meeting every combination of circumstances when using the process to estimate design loads. To begin with the probability of meeting all possible combinations of wave height and period ($P(H,T)$) in the two parameter spectrum will need to be established. It will then be necessary to assess the proportions of operational time spent at different speeds and headings with respect to the wave conditions, noting that courses and speeds will be constrained by severe weather and therefore their probabilities will be functions of wave height. Values of $P(H,T)$ depend on the areas in which the ship will operate and data can be extracted from references such as Hogben et al (1986). The other two probabilities are less easily determined and perhaps the only source of data would be ships' logs, leading to considerable uncertainty over the reliability of the process.

48 Assuming that such information is available, the probability that the peak bending moment B in any one cycle, which can now occur in any of a wide range of conditions, exceeding any given value B is given by

$$(3.35) \quad P(B \geq B) = \int \int \int \int P(V|H) \cdot P(\chi|H) \cdot P(H,T) \exp \left[\frac{-B^2}{2m_{or}(V,\chi,H,T)} \right] d\chi dV dT dH$$

This formula can be simplified somewhat if it is remembered that it has been assumed that responses are directly proportional to wave height. Nevertheless, the numerical computation of the probability is tedious and requires the input of a very large quantity of data, some of which may be of doubtful ancestry, particularly that relating to operational scenarios. The probability of a particular bending moment being exceeded in N cycles can be found from an extension of equation 3.33.

49 Values of bending moment calculated in this way can be compared for particular probabilities of exceedance with values measured at sea and plotted in the Gumbel form, *Figure*

3.4. However, the calculated loads are always significantly greater than those derived from strain measurements, even though slamming is not included in the theory (see paragraph 51), mainly because of the assumption of geometric linearity, use of simple beam theory, sea state assumptions and uncertainty as to the ship condition during strain measurements, and so it is necessary to deduce some form of factor to use to achieve realistic design loads.

Allowance for Hull Flexibility

50 All of the above theory assumes that the hull is acting as a rigid beam. The hull is of course in reality flexible, and it is possible to include terms in the equation of motion for each strip (3.18) which take account of the shear and bending stiffness of the strip and relate shear force and bending moment to the applied forces (Bishop and Price (1979)). Such analysis will yield the flexible modal responses as well as the rigid body responses, and if there are any distortional resonances in the frequency range of interest there will be further bending moments and shear forces to add to those due to direct wave motion. Under normal conditions the flexible responses are sensibly the same as the rigid body responses, but if slam or explosion induced whipping is included then the flexible analysis is essential for a complete understanding of the phenomena. Unfortunately, however, the responses of interest to the structural designer are well into the non-linear range and so current theory can only give a very approximate guide to likely loads or motions and should be used with great caution, but there are occasional incidents, for example those described by Bishop et al (1986) and Lundgren et al (1989), when the flexible theory provides an insight into unexpected happenings which would otherwise not be accessible.

Slamming

51 So far only loads occurring at wave encounter frequency have been discussed. However, waves can also cause loads at much higher frequencies due to impacts between the ship's hull and the water surface, known as slamming. Clearly, this type of response will usually occur when some part of the bottom of the ship comes out of the water and then re-enters, but similar responses can be seen resulting from very rapid changes in added mass and damping due to sudden variations in hull velocity and acceleration. Slamming in theory can happen at any point along the length, but the most critical area will be at the keel in the forward part of the hull. Important slams can also occur at the stern in following seas, and at flare in the bow. The theory and means of load prediction are much the same in each case, so only the most severe case of keel slamming forward will be covered here.

52 The immediate effect of a slam is a short duration high pressure pulse on the hull plating. Typically, at the keel, this pressure may rise to as much as 2 MPa in 0.05-0.1 sec, decaying over a further 0.2 sec. Where relative velocities between water and structure are lower the pressure will be less but the rise time will be greater, up to 1.5 sec. This pressure pulse causes responses at all three structural levels, primary, secondary and tertiary. The primary response consists of resonant vibration of the hull as a beam in all modes of which the lowest three or four will be significant.

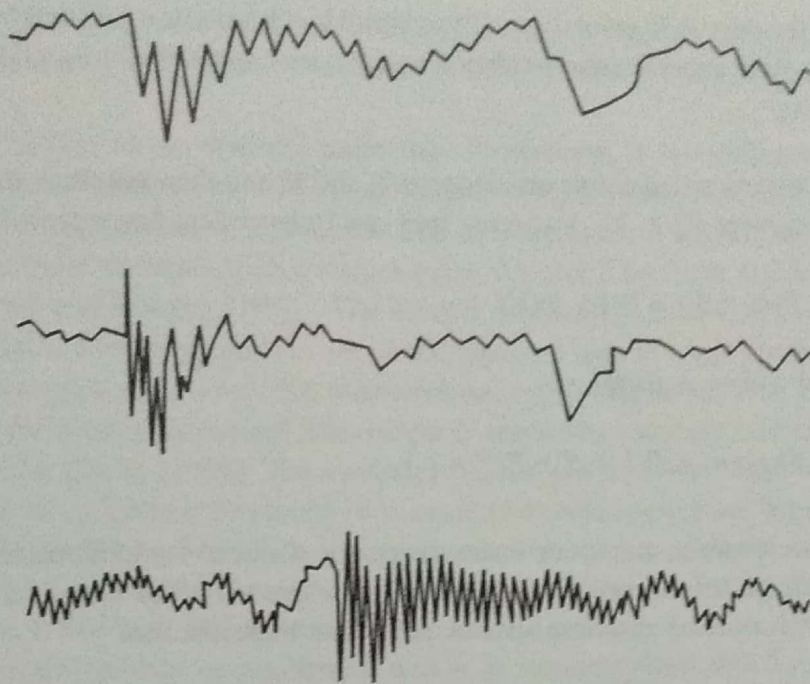


Figure 3.9 Typical Vertical Bending Moment Responses During Slamming

The initial excitation usually occurs as the keel passes through a wave trough when a crest is near amidships and builds up to be directly additive to the wave sagging load as the trough reaches amidships (*Figure 3.9*), leading to a stress enhancement of 50% or more of the normal wave bending stress at amidships, and even more in the quarter of the ship's length forward of amidships (see paragraph 58).

53 The secondary and tertiary responses follow the effect of the pressure on the local plate and grillage structure and can usually be solved in a quasi-static manner when the magnitude of the pressure is known. An understanding of the total response therefore requires the solution of a number of separate problems:

- a. In a particular seaway, the frequency with which slams occur.
- b. The pressure distribution along the bottom of the ship.
- c. The time history of the pressure pulse.
- d. The local structural response to that pressure/time history.
- e. The complete dynamic response of the hull vibrating as a beam (whipping).

54 The two classical works on slamming are by Ochi (1964) and Ochi and Motter (1973), and the theory is briefly summarised by Price and Bishop (1974). Ochi bases the existence of a slam on two conditions being met; that the relative displacement at the bow is such that the bow emerges, and that the relative velocity between the ship and the sea is above a certain threshold

value. Originally the threshold velocity, $\dot{s}_*$, was stated by Ochi to be equal to $0.093\sqrt{Lg}$, but this does not hold for later experimental evidence and there is some doubt as to the real value for different ship types.

55 If the two events noted above are denoted U and V and the draught at the forefoot is T_f , then the combined event $(U \cap V)$, assuming they are independent has a probability

$$(3.36) \quad P(U \cap V) = P(U) \cdot P(V)$$

The probability of a slam is therefore

$$(3.37) \quad P(\text{slam}) = P(U > T_f) \cdot P(V > \dot{s}_*)$$

In the same way as response amplitude operators can be deduced for motions and loads, RAOs can also be found from strip theory or model tests for the two variables T_f and $\dot{s}_*$, and from these may be deduced the relevant response spectra. It follows therefore that

$$(3.38) \quad P(\text{slam}) = \exp \left[- \left(\frac{T_f^2}{2m_{or}} + \frac{\dot{s}_*^2}{2m_{2r}} \right) \right]$$

noting that m_{2r} is the second moment of the response spectrum because $\dot{s}_*$ is a velocity. From this can be deduced the expected number of slams in unit time (N_s) which is

$$(3.39) \quad N_s = (1/\{2\pi\}) \sqrt{(m_{or}/m_{2r})} \cdot P(\text{slam})$$

56 Current empirical methods only allow for an estimate of peak pressure at some point on the keel on normal hull forms. Ochi and Motter (1973), from a large number of wedge drop tests, postulate that the pressure $p = k\dot{s}^2$, where k is a hull form parameter constant for a given section. As the probability distribution of $\dot{s}$ is known then the probability density function of p can be derived, hence the short term probability of p exceeding any limiting magnitude.

57 Ochi's conclusions from his drop tests are:

- a. k is independent of ship speed up to 15 knots for a 150 m ship.
- b. k is independent of sea state.
- c. k is a function of hull form, particularly shape up to 10% of draught from base.
- d. A 'V' form is better than a 'U' form for pressure magnitude.
- e. Peak pressure moves aft with increasing speed.

It should be noted, however, that Ochi *assumed* the relationship in paragraph 56 and all his results are plotted to straight lines on a log scale, and that all the results from ship model tests are lower

than those from 2-D wedges at the same velocity. Further prediction of values of k by Ochi, Lewison and Belkin may be found in the report of ISSC Committee 8 (1973).

58 The time history of the pressure pulse may be explored in two different ways, either or both of which may be input to strip theory, although a flexible beam method is necessary due to higher frequency responses being significant. The first approach is known as *impact slamming* and is due to the short duration, high pressure pulse discussed by Ochi and Motter (1973) and refined by Stavovy and Chuang (1976). The second approach is known as *momentum transfer slamming* and covers a longer period (~ 1 sec) lower pressure load as water is moved outwards due to the downward motion of the ship; this phenomenon is discussed in detail by Leibowitz (1963). Both methods have been incorporated into computer simulations and are compared by Belik and Price (1982) and by Clarke (1986). The computer simulations have been used by Bishop, Clarke and Price (1984) to compare with results of comparative seakeeping trials between two frigates (see Clarke (1982)) and have been shown to give reasonable agreement, but the complexity of the input data required is such that the method is not yet suitable for use during the design process. Clarke (1986) has, however, shown that the effects of slamming on vertical bending moments extend significantly forward of amidships, and it is recommended therefore that the design bending moment lengthwise distribution, as derived either by static balance or by strip theory, is extended forward at a constant value from the maximum at amidships to a point at $0.35L$ from the bow, and then reduced linearly to zero at (or near) the fore perpendicular (see also Chapter 6).

59 With regard to local response, Jones (1973) has shown that in general for monohull type scantlings it is permissible to treat the problem as quasi-static, as the natural frequency of clamped plates is much greater than the effective pressure pulse frequency. Consequently, for flat plates any standard analysis method may be employed, although it is probably best to assume that the plate edges are pinned but free to pull in, as the pressure at any given time will be localised and likely to be acting on only one plate panel. For the response of a grillage the same assumptions on frequency of response will apply, that is a quasi-static solution should be adequate and, because of the localisation of the load, it will usually be sufficient to apply the pressure to each stiffener and associated plating independently of other structure. A simplified method for estimating slamming pressures is given in Chapter 10, together with methods of designing the bottom structure to resist the pressure.

60 The last aspect of hull response to slamming is *whipping*, which may also be caused by other phenomena such as underwater explosions (see also Chapter 4 and Chapter 15). As stated above, whipping is the generalised response of the hull, vibrating as a beam, to any impulsive loading which in the current context is slamming. Dynamic analysis of non-uniform beams is now a fairly standard process, for example Myklestad (1944) for aircraft wings, developed by Todd (1961) for ship applications. It is therefore possible without undue difficulty to estimate frequencies and mode shapes for the dry hull, assuming that sufficient information is available on mass, inertia and stiffness distributions. However, extending that analysis to provide reliable results from impulsive loading while at the same time making proper allowance for fluid added mass and damping is still, for practical purposes, unreliable although the theory exists, for example Bishop et al (1978). (This is the same theory as the computer simulation mentioned above in paragraph 58.) It is still necessary to correlate such analyses with measurements from sea to be sure that unexpected deviations in the results have not occurred in the numerical

calculations and at the same time parameters such as damping are far from fully understood. Consequently, numerical modelling of whipping responses should be treated with caution for the present, and recourse should be made to empirical methods based on measurements from sea. From an overall strength point of view this is not too great a restriction as the effects of slamming are implicitly included in all current strain measurements and will therefore be included in any general correlation procedure.

Green Sea Loading

61 Another effect of slamming, and indeed of proceeding at speed into even moderate seas, is to throw green water onto the deck of a ship. The worst affected parts of the ship will clearly be the forecastle and bridge front where quite high pressures and consequential damage may be experienced. However, green seas can affect any exposed part of the structure and pressures must be allowed for in design. It should be noted also that these pressures may occur at the same time as those due to wind (Chapter 4) as well as wave bending loads and therefore combined load effects must be taken into account.

62 The numerical treatment of pressures due to green seas is non-existent (although frequencies may be computed with some degree of accuracy on the same lines as slamming frequency) and so design pressures are derived from a knowledge of structures that have not failed, and the pressures that would have been needed to fail them. A few measurements have been taken at sea but these are mainly using small pressure transducers which measure only very localised peak pressures and are meaningless for design of structure. Consequently there is a wide variation in design pressures quoted by design authorities and classification societies and the requirements of the latter, for particular parts of ships, may differ by an order of magnitude. Traditionally for RN warships a figure of 50 kPa (100 lbf/ft²) has been used for lack of better information, and this figure finds some support in the work of Yagle (1968) who studied the pressure on decks, side shell and deckhouses on a destroyer model in irregular seas. At no time was a pressure greater than 50 kPa measured. In general, this figure may be too severe, but there is some evidence from recent damage on British frigates that it may be too low for the bridge front and for equipment on the forecastle. The following design figures are therefore recommended for frigate and destroyer type vessels; for smaller and larger ships recourse should be had to Classification Society rules for vessels of about the correct size and speed. All pressures should be used as if they are applied statically over an appropriate area of the structure, for example perhaps any 25% of the bridge front. On decks, while a significant area of plating may be subject to the pressure at one time, only a few stiffeners will be loaded by any single green sea. It is probably adequate, therefore, to halve the load when designing stiffeners because the plating will tend to average out the load.

63 Recommended static green sea pressures for frigates and destroyers:

- a. Bridge front and forecastle mounted equipment - 100 kPa (but see g.)

-
- b. Upper deck forward of the bridge front - 25 kPa at the bridge front, rising linearly to $11.0(\sqrt{L} - T_f)$ kPa (if it is greater than 25 kPa) at the extreme fore end where L (LWL) and T_f (draught at forefoot) are in metres.
 - c. Side shell between waterline and upper deck - 50 kPa.
 - d. Exposed areas of the upper deck and 01 deck - 25 kPa.
 - e. Exposed decks from 02 upwards - 15 kPa.
 - f. Exposed vertical surfaces other than the bridge front - to vary linearly from the pressure required for the contiguous side shell or deck at the bottom to that required for the deck over.
 - g. High value equipment, for example weapons in exposed locations such as the forecastle, may need to be designed to withstand up to 200 kPa to minimise the risk of loss or unacceptable degradation.

Horizontal Bending and Torsion

64 Both horizontal bending and torsion are caused by wave action when seas off the bow are being met. Both can be given the same type of analytical treatment as vertical bending except that the anti-symmetric degrees of freedom will be needed. An example of use of the theory for these responses, albeit for higher frequency distortional modes, is given by Bishop et al (1986). However, for conventional monohull vessels, stresses due to both these responses are much lower than for vertical bending and can usually be ignored, especially as their maxima are likely to occur when the vertical response is well off maximum, although it has been shown on a trial that some vertical and lateral responses in *moderate* seas can be in phase with a ratio 2.5:1. An exception may need to be made for ships with very large deck openings (for example container ships and possibly warships, if the machinery downtakes and uptakes are particularly large) and for ships of very shallow section when torsional stresses may be significant.

65 There is very little available information on which to base estimates of wave induced torque on ships. The most recent is that suggested by Meek et al (1972) who propose for the OCL container ships a figure of $0.0055LBD^2$ tonne.metres maximum at amidships with a (1-cosine) distribution lengthwise. However, there is some evidence from strip theory that rolling has a significant effect on torsional loads and that the maximum could be reduced slightly but should be extended some distance either side of amidships producing a trapezoidal form of distribution. A good compromise would appear to be a maximum torque of $5 \times 10^{-5} LBD^2$ MNm over the centre $0.6L$ tapering linearly to zero at the ends, but further discussion and other proposed formulae are presented by Vedeler (1924) and De Wilde (1967) together with some experimental verification by Lloyd et al (1980).

66 Horizontal bending (that is bending about a vertical plane through the centreline) is usually the least significant of the wave load effect, but it will also occur automatically as a coupled response with torsion because of lack of symmetry in the horizontal plane and lack of correspondence between the shear centre and centre of area of structure, and also due to rudder forces during turning. Very little work is available which is of use to the designer but Elbatouti et al (1974) working on container ships have shown that the worst condition is with waves of length $L/2$ at 60° off the bow. The maximum bending moment can be as much as 75% of the vertical bending moment but is usually much less. Consequently, especially as for conventional designs, the bending strength of the hull is greater horizontally than vertically due to B being greater than D , it is generally safe to ignore horizontal bending. However, if there are large deck cut-outs which require consideration of torsion the additive effect of horizontal bending should be allowed for, either using approximate estimates based on the above references or from model or full size tests. Some more recent work on beam modelling for torsion and bending is reported by Pedersen (1983) but is not yet of any great help to the designer.

Transverse Loads

67 Only those transverse loads due to wave action will be considered here as they apply generally over the surface of the hull shell. Local loads, for example from berthing, will be covered in Chapter 4.

68 The lateral forces resulting from sea pressure in waves are functions of draught and trim, wave characteristics and ship motion. Any load prediction method allowing for all these factors would be complex and, although theoretically possible, would be too time consuming for practical use. For warships therefore a scheme of design pressures has been deduced based on studies of existing structure, and estimates of loads that would have been necessary to damage the structure, so as to provide upper bound load levels. The philosophy is similar to that applied to green sea loading but is *less* soundly based.

69 The loads have two components consisting of:

- a. A uniform pressure head of 5 m representing the dynamic pressure resulting from the orbital velocity of the water, and
- b. A linearly varying head allowing for motion of the ship relative to the sea surface.

Both these heads are present simultaneously but are not additive, the larger of the two being the design figure at any point. The second of the two components should only be applied to one side of the ship at once, as it is intended to take account of the effects of rolling; the significance of this restriction is that decks do not suffer in-plane compression as a result of the load (the out-of-balance force causes cyclic lateral accelerations). A graphical representation of the loads and their magnitudes is given in *Figure 3.10*. It should be noted that these loads do not allow for slamming or green sea pressures which are discussed above.

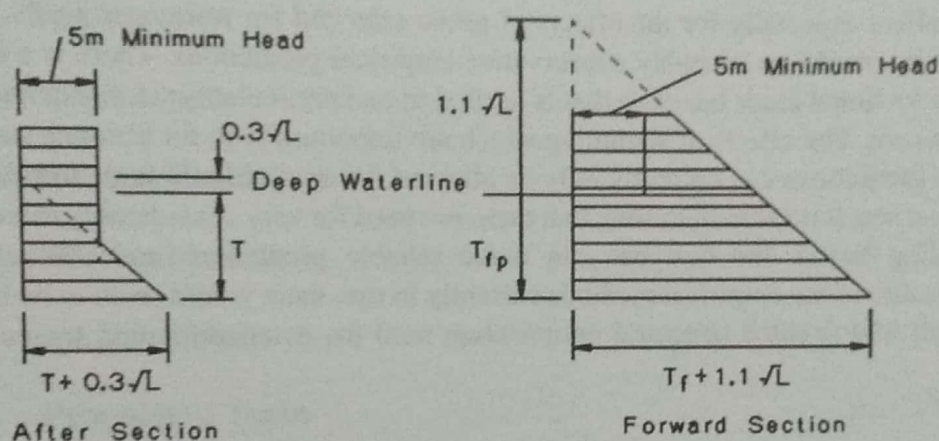
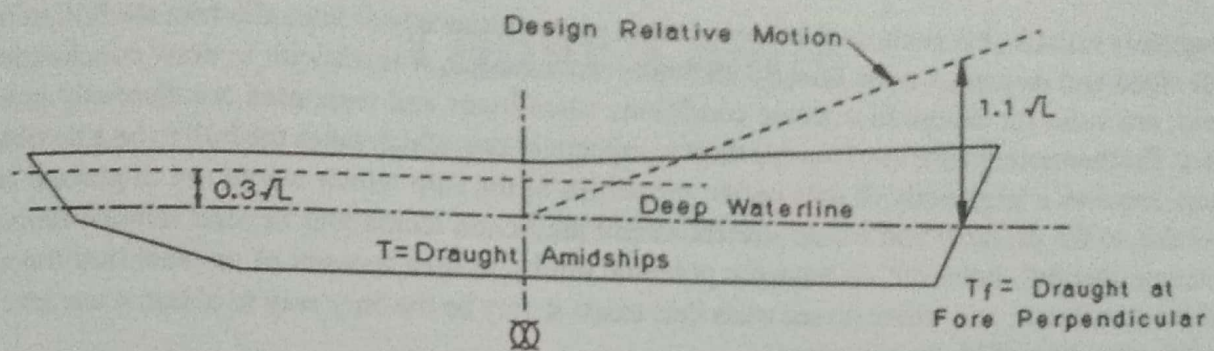


Figure 3.10 Hydrostatic Heads Recommended for the Design of the Bottom and Sides

Conclusions

70 An extensive literature, both historical and contemporary, exists in support of quasi-static estimates of design wave bending moments. The majority of these, however, relate to some arbitrary severe, but not extreme, condition which fits neither a serviceability nor an ultimate limit state. The ultimate limit state which is now recommended for design and which will compare with ultimate strength estimates to follow (Chapter 7) is covered by an update of the quasi-static estimate which allows implicitly for slamming and other potential inaccuracies by means of correlation with measurements from sea and with corresponding partial safety factors to generate a design bending moment with a known probability of exceedance in a given number of wave encounters. This approach allows for non-linearities in both ship and wave form but can only allow for the dynamic effects of motion implicitly through the correlation with measurements. Consequently, it cannot take account of novel or unusual hull forms and, while different operational scenarios can be allowed for in the mathematical model, any extrapolation from the existing database must be treated with great caution.

71 Conversely, in recent years a great deal of effort has been put into dynamic analysis of ship response to waves using strip theory, both for motion and for loads, and allowing for the flexibility of the hull. For moderate sea conditions and responses the results of strip theory are

adequately reliable, but because the theory makes assumptions which inter alia take the hull to be wall-sided and responses to be linearly related to wave height, it is difficult to draw conclusions which are valid for design in extreme conditions when loads and responses are distinctly non-linear. Furthermore, to use the dynamic theory, especially that which takes the hull to be a flexible beam, requires a large body of data on the properties of the ship which would not in general be available to the designer and would stretch current prediction techniques beyond reliable limits. Dynamic theories, however, do have the potential ability to take account of unusual hull forms and sea conditions, and where no sea trials data exists it may be the only way to obtain some level of understanding of likely responses.

72 For the other loads applied to the ship by the passage of waves there is a dearth of reliable design information, especially for the effects of green seas and for transverse loads. For these reliance has to be placed on probably conservative empirical predictions. There is a little more information on torsional loads but even that is limited to one particular type of ship and needs to be treated with care. The effects of slamming which are important both for ultimate strength and for fatigue life predictions can currently only be allowed for implicitly through correlation with measurements at sea. It is clear, therefore, that there is a need for very considerably more research into hull loading before the designer can make reliable predictions and, because of the conservative nature of the empirical methods currently in use, there would seem to be little point in putting effort into detailed structural optimisation until the detailed loading regime is better understood.